

Development of a new concept solar-biomass cogeneration system



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ABSTRACT

A new concept of a system based on a Stirling engine for the combined production of heat and electric power is presented. The system uses two renewable energy sources, direct solar (thermodynamic solar) and biomass (indirect solar energy). Biomass combustion is conducted using a fluidized bed combustor. A second source of energy, given by the direct irradiation of the bed with a concentrated solar radiation, is integrated in the same system, using the fluidized bed as solar receiver. A Scheffler type mirror is adopted to allow irradiation of the system in a fixed focal point. A Stirling engine, integrated into the fluidized bed, converts heat into electricity. Advantages of the proposed solution are illustrated and some preliminary results on the performance of the system, obtained with a simple model, are presented.

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1. Introduction

This paper presents the first developments of a new cogeneration system driven by two renewable energy sources: direct solar (thermodynamic solar) and biomass (indirect solar energy).

There are several technologies available for cogeneration: Roselli et al., in [1], focused on microcogeneration systems (electric power ≤ 15 kW), which represent a valid and interesting application for residential and light commercial users. The energy, economic and environmental performance of small scale cogeneration systems were reported, by means of experimental research activities performed by the authors and other researchers. In [2], the technical performances of complex small scale trigeneration systems are analyzed, through theoretical and experimental investigations. These systems can be based on different prime movers (Stirling engine, reciprocating internal combustion engine, fuel cell, gas turbine) and can achieve significant primary energy savings and reductions of equivalent CO₂ emissions with respect to conventional system, based on separate energy production. In [3], a comparison of various cogeneration configurations, such as vapor and gas turbines, internal and external (Stirling, Ericsson) combustion engines, is performed considering different thermodynamic criteria, namely the energy and exergy efficiency. Optimization of these systems is carried out to maximize the exergy

efficiency, when various practical or physical constraints are taken into account. The gas turbine engine-based one is considered the more representative system for industrial applications of cogeneration. As regards residential and light commercial applications, the Stirling engine (SE), in particular if activated by a renewable energy source, appears the most suitable and promising for small size systems. Microcogenerators (MCHP, micro-combined heat and power, with electric power lower than 15 kW) based on Stirling engine, having electric efficiency in the range between 10% and 25%, and fueled with fossil fuels, are nowadays commercially available [2,4].

The coupling of a biomass system with a Stirling engine to obtain a cogeneration system is therefore quite natural and indeed already exploited in the past.

The original idea, successfully adopted in several systems already on the market, is to place the hot head of the SE in direct contact with the flame, identified as the region of highest temperature, to maximize thermal exchange between the hot combustion gases and the working fluid of the engine. While this is an effective solution for gas combustors, it is not suitable for biomass. Therefore, in usual designs of biomass fueled SE, heat is recovered from exhaust gases, while combustion of biomass takes place in fixed-bed systems, such as gratings typically used to burn logs and wood chips, or braziers used for combustion of pellets [5]. A major drawback of this concept is that, even in the cleaner burners that can be imagined, it is difficult to prevent formation of solid particles that deposit on heat exchange surfaces, progressively reducing the efficiency of the heat recovery [6].

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A	area of the FBC section, m^2
A_H	exchange area of the SE heater, m^2
A_L	exchange area of the SE cooler, m^2
A_{FB}	area of the FBC lateral walls, m^2
A_S	mirror area, m^2
c_{pg}	specific heat of gas phase, $\text{kJ}/(\text{kg K})$
c_F	specific heat of fuel, $\text{kJ}/(\text{kg K})$
c_s	specific heat of solid phase, $\text{kJ}/(\text{kg K})$
d_p	particle diameter, μm
h_{air}	enthalpy of air, kJ/kg
h_{out}	enthalpy of hot gases leaving the FB, kJ/kg
h_F	sensible enthalpy of fuel, kJ/kg
I_S	solar irradiance, W/m^2
k_0	heat leak coefficient of SE, W/K
K_H	heat transfer coefficient between the fluidized bed and the SE heater, $\text{W}/(\text{m}^2 \text{K})$
K_I	heat transfer coefficient of the insulating layer of the FBC, $\text{W}/(\text{m}^2 \text{K})$
K_L	heat transfer coefficient between the working fluid of the SE and the cooling fluid, $\text{W}/(\text{m}^2 \text{K})$
M_1	regenerator time constants of compression, s
M_2	regenerator time constants of expansion, s
$\dot{m}_{air}$	mass flux of air, kg/s
$\dot{m}_F$	mass flux of fuel, kg/s
m_s	total mass of solid particles, kg
n	number of moles of the working fluid of Stirling engine, [–]
P_{SE}	useful mechanical power produced by the SE, kW
$\dot{Q}_{C,in}$	heat entering the system with air, kW
$\dot{Q}_{C,out}$	heat losses by convection, kW
$\dot{Q}_{comb}$	heat released by the biomass combustion, kW
$\dot{Q}_{CSP}$	heat collected from the solar concentrator, kW
$\dot{Q}_{D,out}$	heat losses by conduction, kW
$\dot{Q}_{F,in}$	sensible heat entering the system with biomass fuel, kW
$\dot{Q}_{SE}$	heat transferred to the Stirling engine, kW
$\dot{Q}_{th}$	prescribed thermal power output, kW

γ	specific heat ratio of the SE working fluid, [–]
ΔH_F	lower heating value of fuel, kJ/kg
ϵ_R	efficiency of the regenerator of SE, [–]
η_c	combustion efficiency in the FBC, [–]
η_{CSP}	solar collector efficiency, [–]
η_{SE}	Stirling engine efficiency, [–]
η_{th}	FBC thermal recovery efficiency, [–]
λ	compression ratio, [–]
φ	see Eq. (13), [–]
χ	SE temperature ratio, [–]
ρ_g	gas phase density, kg/m ³

CSP	concentrated solar power
CV	control volume
FB	fluidized bed
FBC	fluidized bed combustor
MCHP	micro-combined heat and power
PDR	parabolic dish reflector
Rec	heat recovery unit
SE	Stirling engine

A device for cogeneration sized for household energy demands was developed in the USA by External Power LLC [12]. It consists in coupling a burner for biomass in the form of pellets, with a Stirling generator, and it is potentially able to meet the energy requirements of a single-family housing unit. A German manufacturer produces a cogeneration unit with 3 kW electric and 10.5 kW

As regards the use of Stirling engines activated by solar energy, except than few studies analyzing systems with flat plate collectors, [15,16], nowadays the technology is limited to Parabolic Dish Reflector (PDR); they are concentrating solar collectors, consisting of a reflective paraboloid, equipped with a biaxial tracking system, which focuses the solar radiation on the receiver, placed in the focus of the paraboloid. Solar energy is absorbed by the receiver

and converted to thermal energy; then it is transferred to a heat transfer fluid. Thermal energy can be directly used in the receiver, to generate electricity by means of an electric generator typically activated by a Stirling engine, directly located in the receiver of the system. Alternatively, thermal energy can be transferred to a centralized system, also in this case for electric energy “production” by means of a Stirling engine.

PDR systems do not have a thermal storage capacity but they can be hybridized to operate with fossil or renewable fuels when solar radiation lacks. To the authors’ knowledge, only few studies have been developed up to know to investigate the feasibility of this coupling [17]. This is one of the aims of the project illustrated in this paper.

Specifically, we are attempting to introduce the fluidized bed technology as a mean able to increase the transfer rate of both the concentrated solar heat and the heat produced by biomass combustion to the Stirling engine.

The main motivations supporting integration of the Concentrated Solar Power (CSP) are: the large and unconditioned availability of the solar energy, especially in tropical and subtropical regions; the possibility to greatly reduce the consumption of biomass, that becomes limited to the supply of the primary energy required to maintain the energy output at nominal levels of power and efficiency during night and/or reduced solar radiation; the possibility to shorten the economic payback time. Furthermore, the possibility to store heat in the inert material of the bed allows to dump some of the variability intrinsic in the uncertainty of atmospheric conditions. This allows to obtain, besides fuel flexibility and low emissions, typical of fluidized bed systems, also optimal operating conditions for the Stirling unit, as it becomes independent from the aleatory solar source and it can reach a high overall efficiency in each operating mode.

The aim of this paper is therefore to describe, model and verify, by means of simulations, the technical feasibility of the new concept of solar-biomass cogeneration system. This will allow to define the optimal configuration of the main components, in terms of technical and dimensional characteristics, towards the maximization of the overall energy performance. In fact, as regards dimensional issues, potential applications of the system in urban environments are mainly represented by residential (both single family and multifamily houses), tertiary and light commercial users, therefore its size should be enough low to be easily located within the building premises. Furthermore, as regards energy performance, the maximization of thermal recovery from the system is particularly advantageous and should be strongly pursued, as it can be used for heating and domestic hot water production purposes, [18–21], in residential and commercial buildings.

Future developments will be the experimental testing of the prototype cogeneration system, for model validation purposes, as well as its comparison, in terms of energy and emissions performance, with a suitable reference system, e.g. based on the separate “production” of the same amounts of electric and thermal energy.

2. System description

The system here reported is sized with the objective to satisfy the needs of a single family residential unit.

The first innovation here proposed is the use of a Fluidized Bed (FB) combustion process that offers, at least in principle, several advantages:

- FB combustion is conducted at temperatures of about 850 °C and then it is perfectly compatible with those required for optimal operation of the SE [22].

- At this temperature, emissions of nitrogen oxides are low, making it a favorable mode of combustion in terms of environmental impact.
- Coefficients of heat exchange between the fluidized bed and the surfaces of the heat exchangers immersed in it are, typically, at least 10 times greater than those between a gas and the heat exchanger. Depending on the design targets, this may reduce the size of heat exchange surfaces, and thus the footprint of the equipment, or ensures the required heat flux at lower combustion temperatures, thus reducing heat losses in the flue gas.
- The FB automatically exerts a cleaning action on the surfaces of the heat exchanger, thus maintaining the exchange coefficients stable over time at the design values, minimizing the need for maintenance [23].

A second innovation proposed is to integrate, in the same equipment, a different source of renewable energy, given by the direct irradiation of the bed with a concentrated solar radiation. The FB offers several opportunities to be exploited in conjunction with CSP. Firstly, by considering the FB as a particle laden flow, the receiver can be conceived on the basis of a radiative transfer from the solar radiation to the small solid particles, in principle much more efficient than any convective or conductive heat exchange [24]. Secondly, the heat capacity of the mass of the solid particles forming the bed can act as a dumper of the energy supply oscillations due to the natural characteristics of the solar source. On the other hand, CSP integration implies higher design complexity and cost, and increases the footprint of the whole equipment.

Due to the size, weight and operating temperature of the FB-SE system, a proper choice of the solar concentrator is required. In conventional concentrated collector types, such as parabolic trough, linear Fresnel, parabolic dish and heliostat field collectors, some features that make them unsuitable for this project exist. In order to provide thermal energy to the FB a high temperature heat transfer fluid is complex to manage and at same time one cannot imagine placing the FB in the focus of the concentrator as it cannot be moved freely (it must necessarily be in vertical position) and easily (because of its weight); finally the generous sizes of FB can prevent a significant amount of solar radiation from reaching the bed.

All these limitations can be overcome by the Scheffler concentrator [25], that is suitable for a static target/focus such as that required for the heavy FB-SE assembly. The most numerous applications of this type of concentrator fall within the scope of the solar cookers [26], but the exergetic approach, proposed in [27], demonstrates that Scheffler collectors are more appropriate for industrial application than for cooking.

Sardeshpande and Pillai [28] compared the economic benefits of a Scheffler and a dish concentrator with Fresnel arrangement for medium temperature applications in India, deriving that the former have a greater feasibility and a shorter pay-back period.

Ruelas et al. [29] developed a simulation model of a Scheffler reflector directly coupled to a Stirling engine. This model is based on geometrical and optical considerations relatively to the concentrator and incorporates also a thermal model of the receiver. They validate the model and numerically demonstrate that this configuration can obtain an increase of 7% in thermal efficiency, with respect to a parabolic dish collector.

A very interesting configuration was found in [30] where a 8 m² Scheffler collector is used to heat a distillation system employed for the treatment of medicinal and aromatic plants.

A daily motion around the equatorial axis inclined at an angle equal to the local latitude (as the typical mounting of telescopes for amateur astronomers) is firstly required, carried out by a stepper motor. The Scheffler focus lies on the projection of the daily rotation axis. A seasonal correction is also required by changing

the parabolic shape in an adaptive way, both by rotating the declination angle ($\pm 23.5^\circ$) and by adapting the curvature of the parabolic surface in order to refit the focus on the target area.

A representation of the whole system is schematized in Fig. 1, while a flow diagram of the principal energy fluxes is shown in Fig. 2. It consists of the following main parts: a solar collector, i.e. a mirror for the capture and concentration of the solar radiation; a FB, that plays the multiple role of: concentrated solar energy receiver, heat exchanger with the head of the SE, biomass combustor; a SE, to convert the heat collected in the FB into mechanical and then electrical power, and a heat exchanger to recover the unused low enthalpy heat for heating purposes. Despite the seemingly complex system, all parts can be easily manufactured to be robust and reliable, the most complex being the SE which, as well known, is considered among the most reliable engines. All other parts are inherently robust and characterized by the need of very low maintenance.

In this paper some particular aspects of importance for the correct approach to the design of the proposed scheme are analyzed in detail. Economic aspects are beyond the scope of this paper and will be analyzed in a separate work.

A preliminary sizing of the main component, i.e. the Fluidized Bed Combustor (FBC), is attempted, starting from selecting few design parameters. The first is the global thermo/electric power output, assumed in the range of 18–24 kW, to satisfy the typical demand of a single household. A proper amount for the electric share can be considered between 1 and 2 kW and it is characterized to be almost continuously required, especially in the case of grid connection, whereas the thermal share is subject to a discontinuous request for sanitary uses and to a seasonal request for heating.

The preliminary sizing of the surface of the mirror for solar caption depends on the average solar irradiance, here assumed equal to 800 W/m^2 , lower than the conventional value of 1000 W/m^2 to account for the effective solar energy collected and/or unfavorable climate conditions. Two values for the total radiation need to be considered. The first is related to the thermal power requested by the SE, the second is related to the global thermodynamic efficiency of the SE cycle that determines the power available for electric output. To generate about 1 kW of the electric power, the solar energy flux to collect is a function of the SE efficiency η_{SE} , this last assumed equal to 0.18 as a first approximation. Therefore, if the thermal energy recovery in the system can be performed with an efficiency $\eta_{th} = 0.85$, the required power becomes:

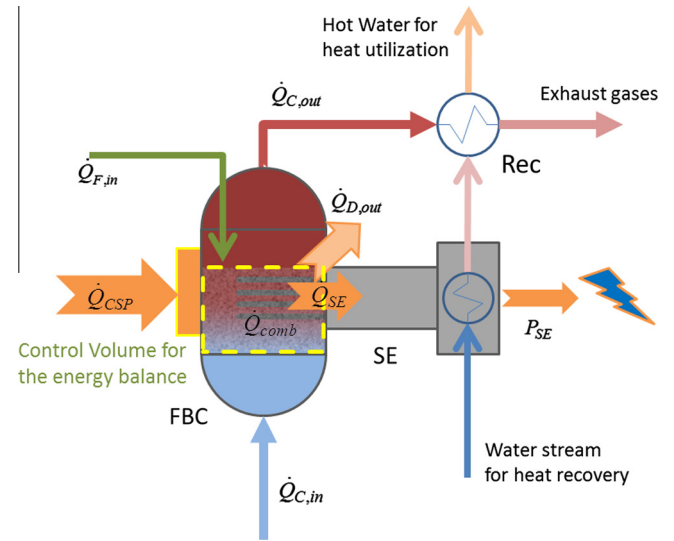


Fig. 2. Flow diagram of the system with the indication of the control volume for the energy balance with the accounted energy fluxes.

$$\dot{Q}_{CSP} = \frac{P_{SE}}{\eta_{SE}\eta_{th}} = \frac{1}{0.18 \cdot 0.85} \text{ kW} = 6.54 \text{ kW} \quad (1)$$

If the solar collection efficiency is $\eta_{CSP} = 0.75$, then a capturing surface of 10.9 m^2 could be sufficient. In the case that the CSP is used for domestic heating, assumed a thermal power request of 20 kW, the CSP to be transmitted to the fluidized bed results:

$$\dot{Q}_{CSP} = \frac{\dot{Q}_{th}}{\eta_{th}} = \frac{20}{0.85} \text{ kW} = 23.5 \text{ kW} \quad (2)$$

With the same assumptions for the efficiency of the solar collection, a much larger capturing surface of 39.2 m^2 would be necessary. These rough estimates clearly indicate the advantage in coupling, in the same system, a second source for heat generation, still maintaining the option of producing electricity without the supply of an additional energy source. Conversely, the addition of the biomass fuel source allows to get the requested thermal output, maintaining a small size of the solar mirror, while allowing for a continuous energy generation, independently from ambient conditions.

3. Model development

A simple model has been developed to assess the potentiality of directly coupling of a solar energy source and a SE hot heat exchanger directly in the bed of a FBC. It consists of an analysis of the energy fluxes among the several system components.

3.1. Fluidized bed combustor

As anticipated in the Introduction, the fluidized bed reactor plays three roles: concentrated solar energy receiver, heat exchanger with the head of the SE and biomass combustor. The fluidization is obtained, as usual, by air flow blowing from the bottom of a vessel where a certain amount of granular material, like sand, is loaded over an air distributor. This allows all particles to float and move, carrying and efficiently exchanging heat with the other particles, the confining walls and the gas phase. Therefore, inside a FB heat is almost immediately spread to both the gas and solid phase, and temperature can be assumed uniform in space. The high heat capacity of the bed helps avoid oscillations in time of the bed temperature, and therefore the whole process develops in a very

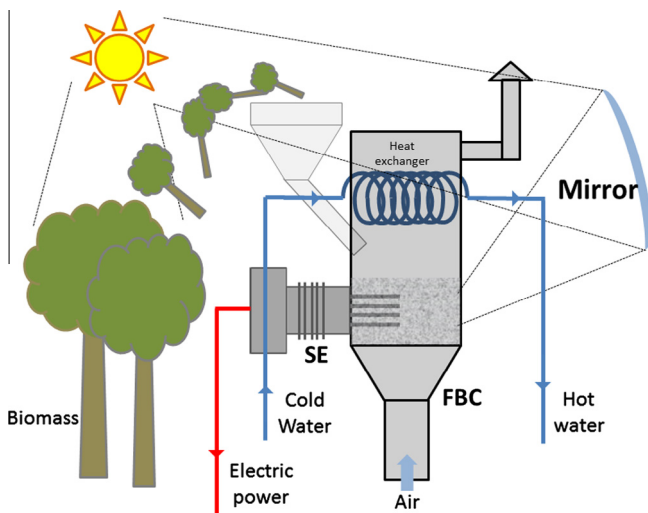


Fig. 1. Schematic of the assembly.

stable and controlled environment, enhancing the performance of the combustion process.

Several models are proposed in the literature to estimate fundamental design parameters for a FBC. Many of these models are based on a 0-dimensional (lumped) approach, with mass and energy balance. This approach is described, with different levels of complexity, in [31–34], and it is here adopted to determine the performance of the system.

Its description starts from the definition of the control volume (CV) used to formulate the budget of total energy, as delimited by the surfaces that surround the volume of the fluidized bed (see the dashed yellow rectangle in Fig. 2). Mass balances are assumed to reduce to a simple global mass conservation, leading to algebraic identity between entering and leaving mass. Vessel walls are assumed to be well insulated, so that only a small amount of energy is lost towards the environment. Radiation flux from the surface of the bed is neglected under the assumption of equilibrium with the reciprocal radiation from the vessel walls. The various flux contributions to the energy budget are identified, as shown schematically in Fig. 2. The energy balance is written assuming the FB working as a pseudo-homogeneous, perfectly stirred reactor including both solid- and gas-phase, meaning that a single temperature is assumed for both solid- and gas-phase. The main energy fluxes are, per unit time:

- The inlet enthalpy carried by the fluidization air, assumed as an ideal gas,

$$\dot{Q}_{C,in} = \dot{m}_{air} h_{air} = \dot{m}_{air} c_{pg}(T_{in}) T_{in}, \quad (3)$$

and having a specific heat value $c_{pg}(T_{in})$;

- The sensible enthalpy carried by the (solid) fuel entering the fluidized bed at temperature T_a ,

$$\dot{Q}_{F,in} = \dot{m}_F h_F = \dot{m}_F c_F T_a \quad (4)$$

and having a constant specific heat c_F ;

- The solar radiation power coming from the collector,

$$\dot{Q}_{CSP} = \eta_{CSP} I_S A_S \quad (5)$$

where η_{CSP} is the global efficiency achieved in the process of collection and transmission to the fluidized bed of the solar power at irradiance I_S adopting a mirror corresponding to a capture surface with area A_S ;

- The thermal power delivered to the SE,

$$\dot{Q}_{SE} = K_H A_H (T - T_1) \quad (6)$$

where K_H is the heat transfer coefficient between the fluidized bed and an immersed surface, evaluated by considering both conductive and radiative terms as reported in [31,35]; it mainly depends on the fluidization regime, the particle diameter and the temperature;

- The outlet power carried by the hot gases leaving the FB, including both fluidization air and combustion gases,

$$\dot{Q}_{C,out} = (\dot{m}_{air} + \dot{m}_F) h_{out} \approx (\dot{m}_{air} + \dot{m}_F) c_{pg}(T) T; \quad (7)$$

- The power lost by conduction from the FBC surface,

$$\dot{Q}_{D,out} = K_I A_{FB} (T - T_a) \quad (8)$$

where K_I is the overall heat transfer coefficient from the bed to the ambient, including an insulation layer, A_{FB} the area of the lateral surface of the vessel in contact with the fluidized bed and T_a the external ambient temperature.

- The thermal power due to biomass combustion,

$$\dot{Q}_{comb} = \eta_c \dot{m}_F \Delta H_F \quad (9)$$

where η_c is the in bed combustion efficiency (see later on), $\dot{m}_F$ the mass feeding rate of fuel, and ΔH_F the lower heating value of the fuel.

Under typical conditions of operation of a FB, the gas velocity is in the order of a few meters per second or less. The bulk gas velocity is computed by assuming that

$$\dot{m}_{air} + \dot{m}_F = A U \rho_g \quad (10)$$

where A is the FBC section, U the velocity of the fluidization air, ρ_g the gas phase density at temperature T . The inlet mass flow rate of air corresponds to the minimum fluidization velocity in the case that no combustion occurs in the FBC ($\dot{m}_F = 0$), while it takes the value corresponding to the combustion air required for a prescribed air excess in the case of combustion. Indeed, the minimum fluidization flow rate is largely overset by the amount of air required to burn the minimum quantity of biomass for practical heating purposes.

Also, the pressure drop needed to fluidize the bed is relatively small. It is possible to neglect the variations of pressure in the bed and assume that the density variations are due primarily to changes in temperature (subsonic flow). In this case $\rho_g T = \text{const.}$ Accumulation of heat in the volume, because of the large mass difference between the solid and the gas, can be assumed to occur only in the solid phase, having total mass m_s and specific heat c_s . Then, the following expression for the total energy balance in the FB is derived:

$$\frac{dT}{dt} = \frac{1}{m_s c_s} [\dot{Q}_{C,in} + \dot{Q}_{F,in} + \dot{Q}_{CSP} - \dot{Q}_{SE} - \dot{Q}_{C,out} - \dot{Q}_{D,out} + \dot{Q}_{comb}] \quad (11)$$

It is worth noting that the thermal power generated by biomass combustion depends on both fuel properties and FB operating conditions. Since biomass fuels are very reactive, a large fraction of combustion enthalpy is released in the bed under usual operation. The remainder fraction is released in the freeboard, contributing to its overheating [36]. In the present model, for the sake of simplicity, the heating of the bed due to heat released in the freeboard region is neglected, while the amount of heat released in the bed is computed by expressing the combustion efficiency η_c as a function of the actual FBC conditions, as reported in [23]. For the adopted fuel properties, corresponding to wood pellet, the efficiency of in bed combustion results to be around 90%.

3.2. Stirling engine

To express the heat power exchanged with the head of the SE when the working conditions inside the FB change, and to compute the corresponding performance, a simplified SE model is required [37]. In this work the SE is modeled adopting the approach proposed in [38]. This model compute the global efficiency of the SE, hence the useful power for electricity production, as a function of all main variables that define the engine and describe the functioning of all its parts, whose values are reported in Table 1.

A detailed description of the model is reported in the original paper [38]. Here, for sake of completeness, only the model equations are reported. The power produced by the thermodynamic cycle of the SE when the working fluid temperatures are, respectively, T_1 inside the expansion chamber and T_2 inside the compression chamber, is given by

$$P_{SE} = (1 - \chi) \varphi(T_1). \quad (12)$$

In this expression $\chi = T_2/T_1 \approx \sqrt{T_L/T_H}$, where $T_H \equiv T$ is the temperature of the hot source, i.e. the fluidized bed temperature, and T_L is the temperature of the cold sink. The $\varphi(T_1)$ function includes some of the most important dependencies upon the SE parameters:

$$\frac{1}{\varphi(T_1)} = \frac{1 + G_1(1 - \chi)}{K_H A_H (T_H - T_1)} + \frac{\chi + G_1(1 - \chi)}{K_L A_L (\chi T_1 - T_L)} + F_1(1 - \chi) \quad (13)$$

Table 1

Values of the main parameters adopted in the Stirling engine model.

Temperature of refrigerant	300 K	Rate of temperature change of regenerator	1.0e–5 s/K
Mass of working fluid (Air)	0.014 kg	Heat leak coefficient	8 s/K
Specific heat ratio of working fluid (Air)	1.4	Compression ratio	2
Working pressure	15e+5 Pa	Radius of absorber elements	0.004 m
Regenerator effectiveness	0.85	Length of absorber elements	0.2 m
Number of absorber elements	50		

where K_H is the heat exchange coefficient between the FB and the working fluid in the expansion chamber; in the present work, this coefficient is assumed as being limited only by the resistance between the solid granular phase and the external surface of the SE hot-side heat exchanger, or absorber; A_H is the total extension of the external surface of the absorber; T_H is the temperature of the hot heat source, assumed equal to the temperature in the FB, $T_H = T$; K_L is the heat exchange coefficient between the working fluid of the SE inside the compression chamber and the cooling fluid; A_L is the total extension of the external surface of the cooler; T_L is the temperature of the cooling fluid; $G_1 = (1 - \epsilon_R)/[(\gamma - 1) \ln \lambda]$ where ϵ_R is the efficiency of the regenerator, γ is the specific heat ratio of the working fluid, and $\lambda = V_1/V_2$ the compression ratio of the SE; $F_1 = (1/M_1 + 1/M_2)/(nR \ln \lambda)$, where n is the number of moles of the working fluid, R is the corresponding gas constant, M_1 and M_2 the regenerator time constants of compression and expansion, respectively. The thermal power absorbed by the SE heat exchanger from the FB, and the thermodynamic efficiency of the SE, are defined, respectively as:

$$Q_{SE} = [1 + G_1(1 - \chi)]\varphi(T_1) + k_0(T_H - T_L), \quad (14)$$

$$\eta_{SE} = \frac{(1 - \chi)\varphi(T_1)}{[1 + G_1(1 - \chi)]\varphi(T_1) + k_0(T_H - T_L)} \quad (15)$$

where k_0 is the heat leak coefficient between the absorber and the heat sink [38].

This model is not in closed form, as it does not fix the value of T_1 . This temperature can be set by assuming optimal working conditions among those ensuring maximum efficiency or maximum useful power. In any case, this temperature remains close to the temperature of the hot source. For this preliminary analysis, an empirical value of T_1 is assumed which reproduces the maximum temperature difference at $T_H = 1100$ K. The chosen values of all the other parameters, not reported here for sake of brevity, lead to the SE behavior with the bed temperature T_H reported in Fig. 3 for two different value of K_H . The predicted performances at the lower value of K_H , that leads to a heat flux comparable with typical values attainable in existing SE, compare well with the Beale formula [39].

4. Model results

Results from the model previously introduced, taking into account all heat fluxes, including radiative losses, and assuming a relatively small mirror of 4 m of diameter (12.5 m²), are reported in Fig. 4. Other parameters of the configuration adopted are reported in Table 2.

Three different working conditions have been considered to estimate the contribution given by the burning of fuel with different levels of sun irradiation, i.e. full (sunny day time, 1000 W/m²), 50% (cloudy day time), and total absence of the sun irradiation (as during the night), and assuming a 75% of global efficiency of sun power capture. The distribution of energy shares among the different flux contributes is reported versus the amount of biomass burned per unit time when steady state conditions are reached.

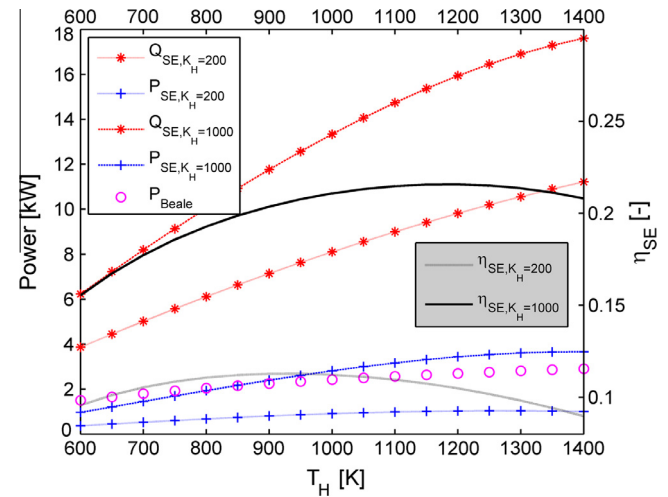


Fig. 3. Mechanical power output, thermal power input and efficiency of the SE model as a function of the fluidized bed temperature T_H at two different values of the hot side heat transfer coefficient. K_H expressed in W/(m² K). With circles the power produced following the Beale's formula.

When all the CSP (yellow line) can be collected, the system is immediately able to produce about 1.25 kW of electricity (blue, continuous and starred line). In this case, addition of biomass combustion helps to increase the power produced by the SE, but increasing the feeding rate the increase of electric production progressively drops and an asymptotic behavior is approximately reached at the feeding rate of 4 kg/s, due to a large increase of the heat transferred to the gas phase. Therefore, in these conditions, most of the energy contribution due to the biomass can be profitably adopted for heating purposes, while it is less justified for the increase of electric energy production.

In this case, the biomass addition is really justified only in the case a much larger request for heating has to be satisfied.

When solar irradiance is halved, the solutions for very low fuel feeding have to be discarded because of the too low temperature reached in the bed to allow steady and clean combustion ($T < 800$ K). In this case, biomass combustion helps to increase the electric output: with 2 kg/h of biomass burned, a significant power output of about 4 kW is reached. Further increase of the fuel feeding is anymore so much effective. When no CSP can be collected, at least 2.5 kg/h of fuels are needed to reach conditions of a bed sufficiently hot to allow steady and clean combustion and to sustain the SE run. These results also indicate that when the combustion of biomass is adopted, the present layout of the system suffers of a rapid increase of energy dissipated by convection (violet line). Indeed, to ensure the proper amount of oxygen for complete combustion, the flux of fed air have to greatly increase well above the minimum fluidization velocity. Contemporary, also the maximum temperature in the fluidized bed increase. While increasing the electric power generated, heat losses also rapidly increase reducing the benefit of a large heat energy production. Conversely, CSP collection greatly helps to reduce biomass combustion, because only small amounts of air need have to be blown

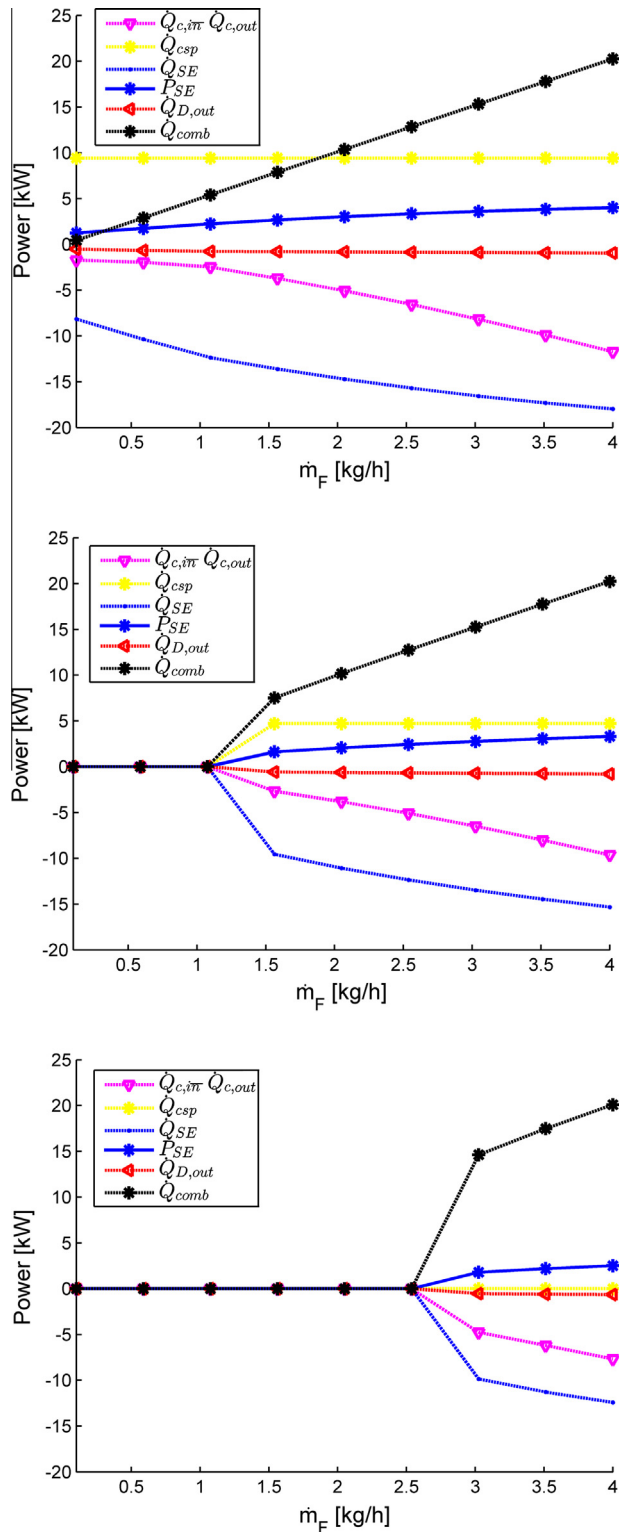


Fig. 4. Energy budgets versus fuel burned per unit time for three different CSP levels of irradiance: (from top to bottom) 1000 W/m², 500 W/m² and 0 W/m². $d_p = 300 \mu\text{m}$.

to ensure the bed fluidization. Several model refinements and design optimizations are still required to get a high level of confidence on the estimated electricity production, but the potentiality of the concept is clearly illustrated.

An interesting comparison can be made by lowering by one order of magnitude the heat transfer coefficient between the bed

Table 2

Values of the main parameters of the fluidized bed reactor.

Reactor diameter	0.300 m	Overall heat transfer coefficient of the reactor	2 W/(m ² K)
Solid particles mean diameter	100, 300 μm	Global CSP efficiency	0.75 [–]
Solid particles d_p	800 J/(kg K)	Lower heating value of biomass (pellet)	18.24e+06 J/kg
Solid particles specific heat	16 kg	Air excess for pellet combustion	30% [–]
Total mass of particles in bed			

and the SE heater, K_H , in conditions of absence of concentrated solar power. This corresponds to imagine the heater located in the freeboard region of the FBC, thus realizing a gas to gas heat exchange, as done in most of the systems already proposed [7–9,11–14]. Keeping the particle diameter to 300 μm , K_H has been then fixed to 20 W/(m² K). Results given by the present model are reported in Fig. 5.

While the possibility to modulate the fuel feeding is enlarged, the electric power output is practically zero, even increasing the fuel feeding to the maximum considered of 4 kg/h. Below the lower limit, not enough heat is produced to warm the bed, while over the upper limit, due to the reduced cooling action of the SE, the temperature very rapidly reaches high values: the entire system reduces to work as a simple stove. These results clearly show that the heater of the SE needs a much larger surface to recover an energy output similar to that previously computed with the heater immersed in the bed. A large SE heater usually reduces the efficiency of the SE, because of the increase of the dead space of the expansion volume. These are probably the main reasons that have inhibited the development of micro-cogeneration systems integrating the FBC and SE technologies, a solution previously explored only for higher power ranges [7–9,11–14] than that proposed for the present system.

The effect of the solid particle size is investigated in Fig. 6. Lowering the size of the solid particles, which corresponds to the adoption of an optimal choice of the solid material with respect to the utilization of the solar heat source, i.e. to choose much smaller particles with diameter of 100 μm [40] to obtain the maximum heat transfer from the fluidized bed acting as receiver and the heater of the SE, leads both to decrease the air flux required for the fluidization of the bed and to increase the heat transfer coefficient between the bed and the SE heater, K_H . However the air flow required for the combustion of biomass can easily leads to the approaching of the terminal velocity for the particle suspension: the air flow required for the combustion determine gas velocities that drag the particles away from the bed.

When focus is fixed on the possibility to maximize the solar source, the plots in Fig. 6 (top), shows that with such small particles a significant increase of P_{SE} up to 1.49 kW from the previous value of 1.25 kW, without the contribution of fuel burning, is obtained. The addition of fuel burning is beneficial for the electricity production: at 2.5 kg/h of fuel feeding rate, P_{SE} increases of up to 3.5 kW from the previous value of 3.34 kW. Furthermore, with so small particles, the air flux required to burn more than 2.5 kg/h of fuel implies a gas velocity higher than the terminal particle velocity, marking a strong limit for the working conditions.

Conversely, in the case of absence of solar irradiance, the limitations become too strong for a profitable use of the system. Only a narrow range of useful working condition is detected, with fuel feeding rate between 3 and 3.5 kg/h. At lower values, the combustion efficiency is too low to release enough heat to sustain the

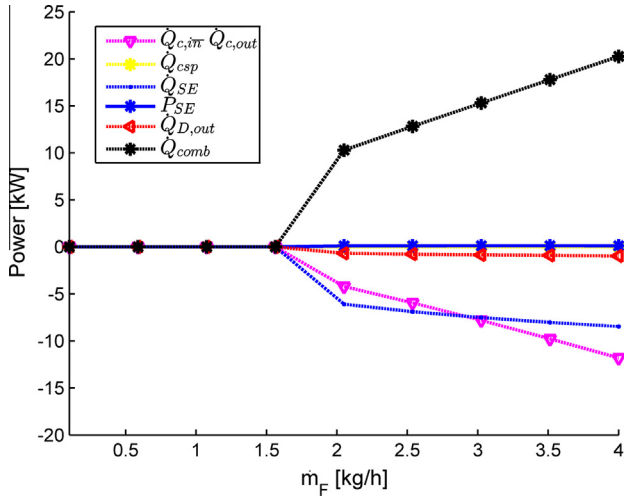


Fig. 5. Energy budgets versus fuel burned per unit time for $K_H = 20 \text{ W/(m}^2 \text{ K)}$ and without CSP irradiance.

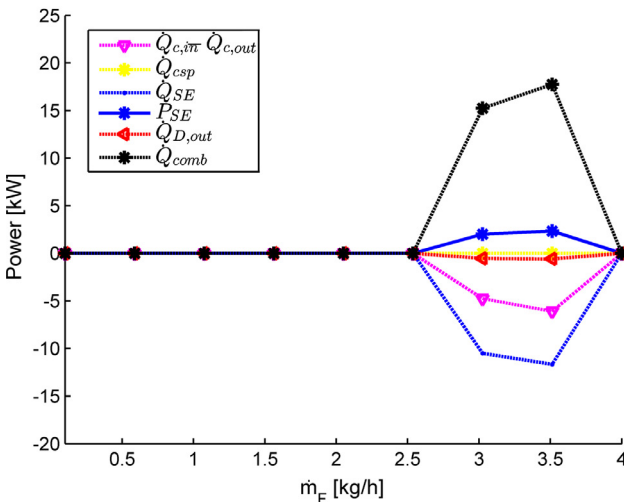
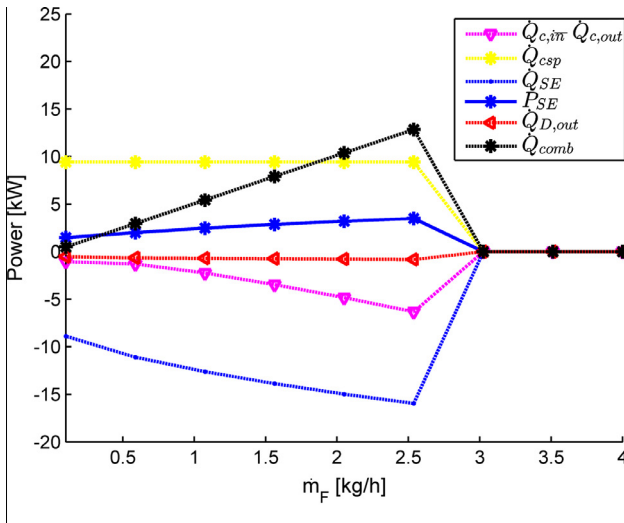


Fig. 6. Energy budgets versus fuel burned per unit time with small particles ($d_p = 100 \mu\text{m}$): (top) at solar irradiance of 1000 W/m^2 and (bottom) without CSP.

combustion. At higher values the terminal velocity of the particles is reached.

5. Conclusions

A new concept for a system that can provide heat and electricity for a single household from renewable energy sources (CSP and Biomass) has been presented. The principal improvements with respect to existing systems of similar size and primary energy source have been illustrated. The most important are the enhanced heat transfer processes that are realized with the help of a fluidized bed and the possibility of continuous cogeneration during day and night. A simple model has been developed and used to evaluate the expected performance. Two main conclusions can be derived from the model results. Firstly, the possibility to adopt a medium, the fluidized bed, ensuring heat transfer coefficients about one order of magnitude greater than that typical of systems recovering the heat from the gas phase, greatly helps in achieving a high utilization of the SE. Secondly, the utilization of the SE adopting the biomass fuel as energy source, is greatly influenced by the size of the sand adopted for the bed. This implies that a careful optimization of the several parameters is necessary to achieve significant amount of electric power produced, otherwise the system perform mostly as a boiler. Activities are currently carrying out, with the help of experimental tests, to assess and improve the flexibility of the system.

Another advantage of the proposed coupling of a FBC and a SE is the possibility to achieve temperatures much higher than those attainable with other media adopted as heat conveyor of CSP, like water vapor or molten salts. This, together with the reduced distance from the point of focalization of the CSP and the SE heat exchanger, greatly helps in attaining the evaluated performances.

One of the main drawbacks of this system is its high investment cost, due to the typical low size that does not allow to benefit from mass-production of its components. Moreover, these systems are still in testing or pre-commercialization phase, with the consequent lack of industrial standardization, in particular as regards structural and optical components. Therefore, even if the technical maturity of the components (in particular the fluidized bed and the Stirling engine) has been achieved since a long time, conversely their commercial maturity is still far from being obtained, and no reliable data about the required investment costs are available at the moment. This does not allow to perform a neither broad economic analysis, in terms of pay-back period.

Furthermore, the system is exactly conceived to perform in continuous operation, thanks to the integration of an unpredictable renewable source (solar) with a programmable one (biomass); therefore, the reduction of energy costs, with respect to a conventional system based on separate "production" of thermal (natural gas boiler) and electric energy (grid), is positively affected by the high number of achievable operating hours. Moreover, in the economic analysis, supporting mechanisms for renewable energy sources (feed-in tariffs, energy efficiency certificates, etc., [41]) should be taken into account as a mean to obtain financial revenues, especially when Energy Service Companies bear the investment costs and the management of the proposed system.

Finally, FB combustors are very flexible toward fuel properties, allowing to use freely available agricultural residues (e.g. chips from trees pruning). This renders the proposed idea more profitable, in particular with reference to a general trend of increasing price for oil and natural gas.

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References

- [1] Roselli C, Sasso M, Sibilio S, Tzscheutschler P. Experimental analysis of micro-cogenerators based on different prime movers. *Energy Build* 2011;43:796–804.
- [2] Angrisani G, Roselli C, Sasso M. Distributed microtrigeneration systems. *Prog Energy Combust Sci* 2012;38:502–21.
- [3] Feidt M, Costea M. Energy and exergy analysis and optimization of combined heat and power systems. *Comp Various Syst, Energies* 2012;5:3701–22.
- [4] Knight I, Ugursal VI. Residential cogeneration systems: a review of the current technologies, A Report of Annex 42 of the International Energy Agency, Energy Conservation in Buildings and Community Systems Programme; April 2005.
- [5] Bain RL, Overend RP. Biomass for heat and power. *For Prod J* 2002;52:12–9.
- [6] Vos J. Biomass energy for heating and hot water supply in Belarus. Contract Report (BYE/03/G31), BTG; 2006.
- [7] Podesser E. Electricity production in rural villages with a biomass Stirling engine. *Renew Energy* 1999;16:1049–52.
- [8] Jensen N, Werling J, Carlsen H, Henriksen UB. CHP from updraft gasifier and Stirling engine. *Eur Biomass Conf* 2002;12:726–9.
- [9] HighBio Project. Biomass Gasification to Heat. Electricity and Biofuels. Ed. Ulla Lassi and Bodil Wikman. HighBio Project Publication; 2011.
- [10] Keck T, Schiel W. Envirodish and Eurodish-System and Status. In: Proceedings of ISES solar world congress; 2003.
- [11] Rogdakis ED, Antonakos GD, Koronaki IP. Thermodynamic analysis and experimental investigation of a Solo V161 Stirling cogeneration unit. *Energy* 2012;45:503–11.
- [12] Tilt, Yumi Koyama. Micro-Cogeneration for Single-Family Dwellings; 2001.
- [13] Thiers S, Aoun B, Peuportier B. Experimental characterization, modeling and simulation of a wood pellet micro-combined heat and power unit used as a heat source for a residential building. *Energy Build* 2010;42:896–903.
- [14] Leu JH. Biomass power generation through direct integration of updraft Gasifier and Stirling engine combustion system. *Adv Mech Eng* 2010;2010:256746.
- [15] Tavakolpour AR, Zomorodian A, Golneshan AA. Simulation, construction and testing of a two-cylinder solar Stirling engine powered by a flat-plate solar collector without regenerator. *Renewable Energy* 2008;33:77–87.
- [16] Kongtragool B, Wongwises S. A review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renew Sustain Energy Rev* 2003;7:131–54.
- [17] Petrescu S, Petre C, Costea M, Malancioiu O, Boriaru N, Dobrovicescu A, et al. A methodology of computation, design and optimization of solar Stirling power plant using hydrogen/oxygen fuel cells. *Energy* 2010;35:729–39.
- [18] Caresana F, Brandoni C, Feliciotti P, Bartolini CM. Energy and economic analysis of an ICE-based variable speed-operated micro-cogenerator. *Appl Energy* 2011;88:659–71.
- [19] Cardona F, Piacentino A, Alterio V. Analysis and optimization of fuel cell cogeneration systems for application in single-family houses. In: Proceedings of 23rd international conference on efficiency, cost, optimization simulation and environmental impact of energy systems, June 14–17, Lausanne, Switzerland; 2010.
- [20] Orr G, Dennish T, Summerfield I, Purcell F. Commercial micro-CHP Field Trial Report, Report of Sustainable Energy Authority of Ireland (SEAI); 2011.
- [21] Carbon Trust. Micro-CHP Accelerator. Interim report; 2007.
- [22] Kim SY, Huth J, Wood JG. Performance characterization of sunpower free-piston Stirling engines. In: Proceedings of AIAA IECEC2005 conference; 2005.
- [23] Miccio F. On the integration between fluidized bed and Stirling engine for micro-generation. *Appl Therm Eng* 2013;52:46–53.
- [24] Kim K, Siegel N, Kolb G, Rangaswamy V, Moujaes SF. A study of solid particle flow characterization in solar particle receiver. *Sol Energy* 2009;83:1784–93.
- [25] Munir A, Hensel O, Scheffler W. Design principle and calculations of Scheffler fixed focus concentrator for medium temperature applications. *Sol Energy* 2010;84:1490–502.
- [26] Kariuki Nyahoro P, Johnson RR, Edwards J. Simulated performance of thermal storage in solar cookers. *Sol Energy* 1997;59:11–7.
- [27] Kumar N, Vishwanath G, Gupta A. An exergy based unified test protocol for solar cookers of different geometries. *Renewable Energy* 2012;44:457–62.
- [28] Sardeshpande V, Pillai IR. Effect of micro-level and macro-level factors on adoption potential of solar concentrators for medium temperature thermal applications. *Energy Sustain Dev* 2012;16:216–23.
- [29] Ruelas J, Velquez N, Cerezo J. A mathematical model to develop a Scheffler-type solar concentrator coupled with a Stirling engine. *Appl Energy* 2013;101:253–60.
- [30] Munir A, Hensel O. On-farm processing of medicinal and aromatic plants by solar distillation system. *Biosyst Eng* 2010;106:268–77.
- [31] Kunii D, Levenspiel O. Fluidization engineering. Boston: Butterworth-Heinemann; 1991.
- [32] Wagialla K, Elnashaie S. Fluidized-bed reactor for methanol synthesis. *A Theor Invest, Ind Eng Chem Res* 1991;30:2298–308.
- [33] Hatzantonis H, Yiannoulakis H, Yiagopoulos A, Kiparissides C. Recent developments in modeling gas-phase catalyzed olefin polymerization fluidized-bed reactors: the effect of bubble size variation on the reactor's performance. *Chem Eng Sci* 2000;55:3237–59.
- [34] Galgano A, Salatino P, Crescitelli S, Scala F, Maffettone P. A model of the dynamics of a fluidized bed combustor burning biomass. *Combust Flame* 2005;140:371–84.
- [35] Molerus O, Burschka A, Dietz S. Particle migration at solid surfaces and heat transfer in bubbling fluidized beds – II. Prediction of heat transfer in bubbling fluidized beds. *Chem Eng Sci* 1995;50:871–7.
- [36] Chirone R, Miccio F, Scala F. On the relevance of axial and transversal fuel segregation during the FB combustion of a biomass. *Energy Fuels* 2004;18:1108–17.
- [37] Mathieu A, Grappe B, Gualino D, Grosu L, Feidt M, Rochelle P. Preliminary sizing and optimization of a micro-solar power plant by a parametric sensitivity study. *Environ Eng Manage J* 2010;9:1381–7.
- [38] Yaqi L, Yaling H, Weiwei W. Optimization of solar-powered Stirling heat engine with finite-time thermodynamics. *Renewable Energy* 2011;36:421–7.
- [39] Kongtragool B, Wongwises S. A review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renew Sustain Energy Rev* 2003;7: 131–15.
- [40] Yang W-C. Handbook of fluidization and fluid-particle systems. CRC Press; 2003.
- [41] Hawkes AD, Leach MA. On policy instruments for support of micro-combined heat and power. *Energy Policy* 2008;36:2973–82.